Investigation of Structural Behavior due to Bend-Twist Couplings in Wind Turbine Blades

Fedorov, Vladimir; Dimitrov, Nikolay Krasimirov; Berggreen, Christian; Krenk, Steen; Branner, Kim; Berring, Peter

Published in:
Proceedings

Publication date:
2010

Investigation of Structural Behavior due to Bend-Twist Couplings in Wind Turbine Blades

V.A. Fedorov*[1], N. Dimitrov*, C. Berggreen*, S. Krenk*, K. Branner¤ and P. Berring¤
*Department of Mechanical Engineering, Technical University of Denmark
DK-2800 Kgs. Lyngby, Denmark
¤Wind Energy Division, Risø National Laboratory for Sustainable Energy,
Technical University of Denmark
DK-4000 Roskilde, Denmark
[1]Corresponding author: vlf@mek.dtu.dk

Summary

One of the problematic issues concerning the design of future large composite wind turbine blades is the prediction of bend-twist couplings and torsion behaviour. The current work is a continuation of a previous work [1,2], and it examines different finite element modelling approaches for predicting the torsional response of the wind turbine blades with built-in bend-twist couplings. Additionally, a number of improved full-scale tests using an advanced bi-axial servo-hydraulic load control have been performed on a wind turbine blade section provided by Vestas Wind Systems A/S.

In the present work attention was aimed specifically at shell element based FEA models for predicting torsional behaviour of the blade. Three models were developed in different codes: An ANSYS and ABAQUS model with standard section input and an ANSYS model with matrix input. All models employed the outer surface of the blade cross section as the defining surface, off-setting the location of the shell elements according to the specified thickness.

The experimental full-scale tests were carried out on an 8 m section of a 23 m wind turbine blade with specially implemented bend-twist coupling. The blade was tested under considerably larger load levels compared to earlier tests and showed linear-elastic response during flap-wise bending and combined bending-torsion tests which made it possible to employ the principle of superposition to extract the torsional characteristics of the blade from these tests. Additionally, pure torsion tests were carried out on the blade employing a more advanced bi-axial servo-hydraulic load application control.

The use of shell-solid models for the prediction of torsional response was recommended based on earlier investigations. However as these models in practise are cumbersome to apply in design, the numerical models mentioned above were compared with previous experiments and the new experiments presented in this paper. Additionally, the models were verified against two older MSC.Nastran models developed in. All shell models performed well for flap-wise bending, but performed poorly in torsion with deviations in the range of 15 to 35%, when employing the section input for the off-set definition. However, the ANSYS model generated using matrix input for the off-set definition was found to perform adequately.

1 Introduction

For future large MW wind turbines, the problem of reliable prediction of the blade torsion behaviour becomes very important. With increasing blade size, bending and torsional natural frequencies of the blade become closer to each other and when sufficiently close there is a possible risk of flutter even under low wind speed conditions.

The Finite Element Method (FEM) is currently used for modelling complex composite structures such as wind turbine blades. This method is a powerful and universal tool for analyzing composite structures with high degree of complexity.

For a number of reasons shell elements are widely used in modelling of wind turbine blades. There are several approaches for creating shell element models. One is to use the wall mid-thickness surface as a basis and place all the shell elements on this surface. A possible difficulty here is the situation when two walls with different thicknesses are aligned with their top surfaces. Such a situation leads to a discontinuity of the mid-thickness surface. Another approach is to use the outer surface of the blade as a basis and employ offset techniques. However, it was shown [1, 2] that models based on shell elements and exploiting offset techniques have a high uncertainty in prediction of torsional behaviour of composite structures.

The present work is a continuation of earlier work carried out at the Technical University of Denmark [2, 3]. The main objective behind the work is to investigate the accuracy of finite element (FE) shell models with offset for prediction of torsional stiffness of wind turbine blades. In addition to FE models developed in the previous work, three new models based on different approaches and solvers were developed, validated against experiments and verified against the earlier models. All three models use the outer surface of the blade as a basis for the modelling of the geometry. Two of them were developed in respectively ANSYS and ABAQUS with layer-by-layer material data input, while the pre-integrated matrix input is used in the third ANSYS model.

The experimental full-scale tests were carried out on an 8.4 meter section from a 23 meter wind turbine blade with intentionally introduced bend-twist coupling. The blade section was tested under considerably larger load levels compared to earlier tests and showed linear-elastic response during flap-wise bending and combined bending-torsion tests. Additionally, pure torsion tests were carried out on the blade employing an advanced bi-axial servo-hydraulic load application control.

All the models showed good convergence in prediction of bending displacement, while only the MSC NASTRAN FE model based on shell and solid elements and the present ANSYS shell model with pre-integrated matrix input showed satisfactory results with regards to torsion prediction. The remaining models performed poorly with deviations in the range of 15-35%. Thus, use of pure shell element models with offset to the outer surface is not advisable for development of future large wind turbine blades, unless the pre-integrated matrix input is used.

The paper is based on the experimental work carried out in [4] and later additional experiments applying the before mentioned bi-axial servo-hydraulic load application control.

2 Experiments

The experimental part of the study was carried out on a real wind turbine blade section. An entire 23 meters length wind turbine blade (Fig. 1) was provided by Vestas Wind Systems A/S, but only an 8.4 meter section (from radial position 10.8 m. to 19.2 m.) was chosen for testing. The most important advantages of the chosen section are: Required slenderness of the section, its moderate stiffness and rather simple geometry. More details concerning preparation of the blade for experiments are given in [5].

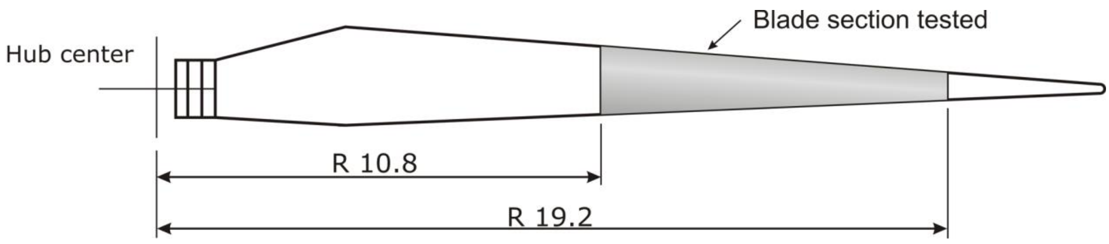


Figure 1. 23 meters wind turbine blade

The blade section is a lightweight composite structure, and consists of fibreglass (E-glass/Epoxy), PMI and PVC sandwich core material and has a protective gel coating on the outer surface. The section consists of two main parts: An aerodynamic outer shell consisting of monolithic and sandwich laminates, and a box-beam consisting of moderate thickness top- and bottom-surface monolithic laminates supported by PMI foam cored sandwich webs. The aerodynamic outer shell consists of thin sandwich panels in the leading edge area and relatively thick sandwich panels in the trailing edge area. The aerodynamic shell and the primary load carrying box beam are glued together with a relatively thick adhesive layer along the contact surface between the two parts.

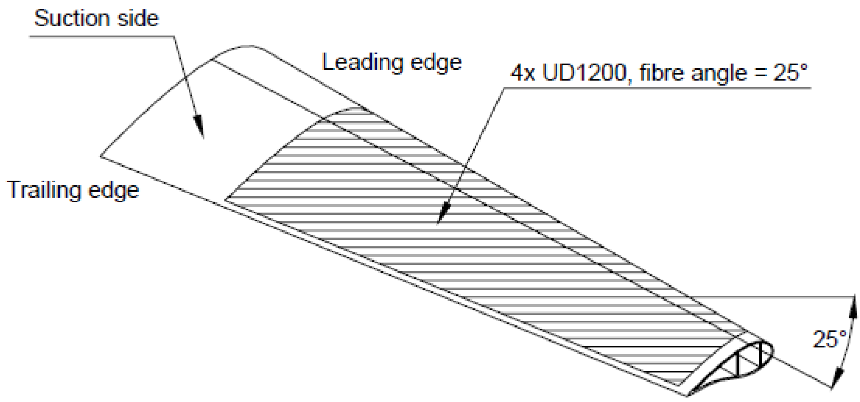


Figure 2. Introduction of bend-twist coupling into the 8.4 meter blade section.

In order to introduce a measurable bend-twist coupling, additional unidirectional (UD) layers were laminated onto the pressure and suction sides of the blade section by a vacuum infusion process (see Fig 2.). The fiber direction for the layers was chosen to 25° with respect to the blade pitch axis [5], based on initial FE analysis.

2.1 Test rig and load application

For the full-scale tests on the blade section clamped boundary conditions were chosen. In order to impose these boundary conditions two root clamps were designed and manufactured [5], fixing the blade by clamps at the root part of the blade section (see Fig. 3). Loads were applied through a third clamp placed at the blade section tip. The loading clamp includes a number of holes designed for the connection with the load application system.

In the present study three different load cases were applied to the blade section. Initially the flap-wise bending load was applied as a 32 kN force pulling upwards at the blade pitch axis position located in the geometrical centre of the box-beam (see Fig. 4a). Next, a combined flap-wise bending and torsion load case was applied as a force of 15 kN pulling upwards at the position located 0.695 meters away from the blade pitch axis (see Fig. 4b).

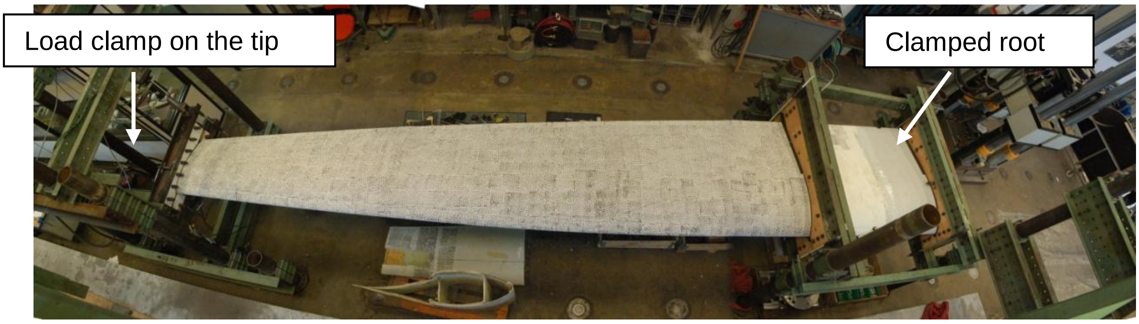


Figure 3. Test rig with the blade section

The force produces a torsion moment of 10.43 kNm in addition to the bending load. Both of the load cases were applied by a crane connected to the load clamp and equipped with a load cell.

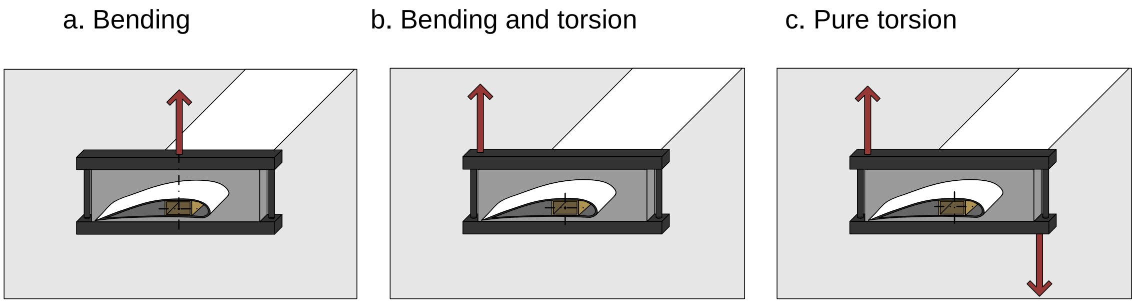
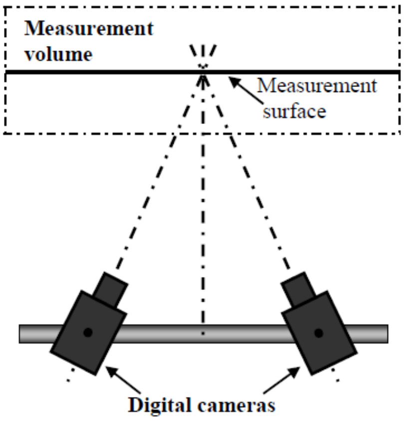


Figure 4. Load cases: a. 32 kN bending force; b. 15 kN bending force and 10.43 kNm torsion moment; c. 4.5 kNm torsion moment.

Finally, an advanced bi-axial servo-hydraulic system with load control was applied to introduce a pure torsion load case. Two actuators were connected to the load clamp with spherical bearings; introducing two equal forces as a force pair, generating a pure and free torsional moment of 4.5 kNm at the tip of the blade section (see Fig. 4c). The applied free torsion moment in this load case is lower than in the bending-torsion load case due to the problem of a growing crack in the relatively soft trailing edge in the area close to the load clamp.

2.2 Displacement measurement system

A Digital Image Correlation (DIC) system ARAMIS 4M was applied in the tests as a non-contact full-field displacement measurement system. The DIC system consists of two spaced and angled digital cameras aimed on the top surface of the blade section. In order for the surface to be recognized by the DIC system, a stochastic pattern of high contrast was applied to the top surface of the blade section (see Fig 5) [6].



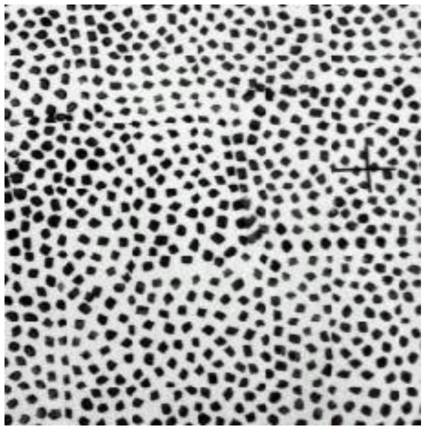


Figure 5. DIC system setup and stochastic pattern applied on the blade top surface.

The digital cameras capture images of the blade surface during the quasi-static testing, and the images are later analyzed during the post-processing stage where the 3D displacement field along the entire blade surface is calculated.

As the blade section is an elongated object the cameras were not able to span the entire free blade length without loosing considerably accuracy, and thus three camera positions along the blade length were chosen to capture the blade surface in three separate parts, which however can be combined in the post-processing stage.

2.3 Determination of experimental deflections and rotations

Analysis of the blade section deformation was performed through the analysis of deflections of a number of cross-sections defined along the entire blade length. As a result of the post-processing stage of the DIC system, the full 3D deflection field of the blade top surface is obtained. Deflections of points along the cross-section curves are then considered.

In order to extract the cross-section bending displacements, planes are fitted to the spatial cross-section curves by a least square algorithm. The rotation of the fitted plane can thus be used to define the bending deflections which are in turn calculated by integration of the rotation angles along the blade length.

For the extraction of the cross-section rotation around the pitch axis, only vertical displacements of the points along the cross-section curve are considered, as horizontal displacements are negligibly small. A line is fitted into the representation of vertical displacements along the cross-section curve by a least square fit. Then the twist angle of the cross-section is considered as a slope of the fitted line.

The calculation of deflections and rotations of the cross-sections is implemented in a series of MATLAB scripts.

3 Numerical modelling

Five FE models of the blade section are validated in the present work. All models use outer surface of the blade as a basis for the geometric modelling. Three of them are based only on shell elements and are almost identical, except that they are developed in three different commercial FE codes. Possible deviations between these models will indicate if the problem of inaccurate prediction of the torsional behaviour by FE models with offset is limited to a particular solver and offset algorithm.

Two MSC NASTRAN models were developed earlier [5] and their performance are compared to three new models. A so-called shell/solid MSC NASTRAN model was built with a combination of shell and solid finite elements. Laminates are modelled with shell elements, while the core material and the adhesive bonds are modelled with solid elements. The model consists of approximately 35,000 Quad8 shell and 45,000 Hex20 solid elements. The additional UD layers are also modelled by solid elements because they introduce considerable change of thickness to the model and this might compromise the model accuracy.

The second MSC NASTRAN model is a pure shell-element model of the blade section. It consists of approximately 10,000 layered 8-node shell elements, which define the blade outer surface and box beam webs.

Two of the remaining three blade section FE models developed in the present work are almost identical to the pure shell MSC NASTRAN model described above. The first is developed in ANSYS while the second is developed in ABAQUS. Both models are pure shell models with geometry representing the outer surface of the blade section with offset and thus geometrically identical to MSC NASTRAN model. The ANSYS model consists of approximately 23,000 4-node shell elements (SHELL181) while the ABAQUS model consists of approx. 23,000 shell elements (S4).

The last FE model developed in the present study is likewise a pure shell model with geometry representing the outer surface of the blade section. It is developed in ANSYS, but instead of straight layer-by-layer material data input and offset to the basis surface like in the two previous models, it is based on pre-integrated sections available in ANSYS, where the material properties are input in terms of the ABD matrices of the laminates. The values of the ABD matrices are obtained from a series of MATLAB scripts using classical lamination theory. Offset to the outer surface in the model is defined by implicitly changing the integration limits through the changing of layer coordinates in the calculation of ABD matrices [7]:

$$[A, B, D] = \sum_{i=1}^{n} Q_i \left[(z_i - z_{i-1}), \frac{1}{2}(z_i^2 - z_{i-1}^2), \frac{1}{3}(z_i^3 - z_{i-1}^3) \right],$$

where A – extensional stiffness matrix, B – extension-bending matrix, D – bending stiffness matrix, Q – lamina stiffness matrix in the global coordinate system, z – coordinate of the current layer and n – number of layers in the laminate.

3.1 Determination of deflections and rotations in the numerical models

For validation of the numerical models against the experiments, deflections and rotations of the same cross-sections as in the full-scale tests are extracted from the results of numerical modelling. Exactly the same calculation algorithm is thus applied as in the experimental post-processing.

However, in the ANSYS models it is possible to exploit the RBE (Rep-bind) elements, which are finite elements using the least square algorithm to extract translations and rotations of a single master node as function of a number of slave nodes. The master node of the RBE element of a cross-section is located at the pitch axis of the blade and connects to all the constituent points of the cross-section. Thus, there is no need to apply the above mentioned algorithm as cross section rotations, deflections and twist angles are obtained explicitly from master nodes

4 Results

As a result of the experimental and numerical parts of this study, bending displacements and twist angles along the blade section are represented for each load case. The same coordinate system associated with the blade section is used for the representation of both numerical and experimental results. It has the plane containing the X (horizontal) and Y (vertical) axes in the plane of the root clamp and the Z axis of the coordinate system coincides with the blade pitch axis.

Eleven cross-sections are assigned along the entire blade section. The cross-sections have a distance of 0.5 meters between them, except the first and the last section, for which the distance is 0.3 meters. The choice of the location of the first and last cross-sections is limited due to the extent of the measured area. However, in the pure torsion tests it was possible to extend the measured area so that also half a meter distance between the last and the second last cross-sections exists.

In order to reduce the effect of the simplified boundary conditions in the numerical models, the results are reduced to the first cross-section in the position of 0.7 meters. Thus, this cross-section will always haves a rotation angle, bending displacement and twist angle equal to zero.

Below (Figs. 6-9) the results are presented for the bending, bending-torsion and pure torsion load cases.

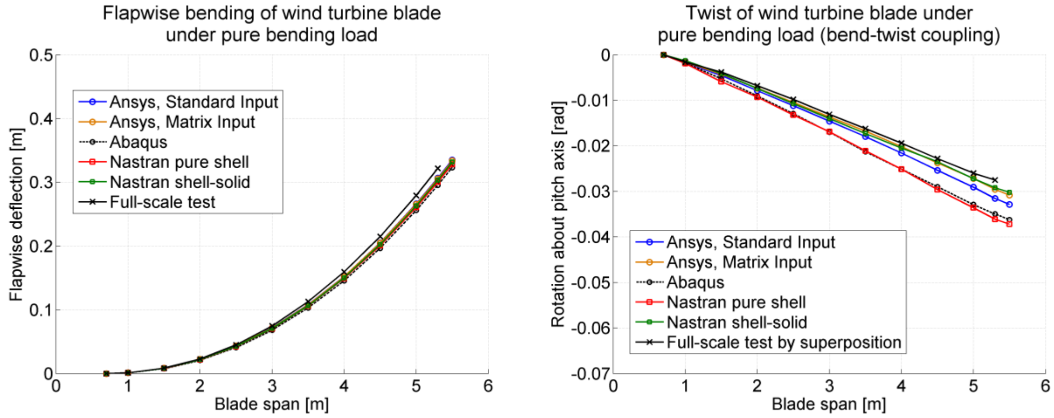


Figure 6. Results of the pure bending load case. Bending force is 32 kN.

In the bending load case (Fig. 6) all the models predict bending displacement well, while the torsional stiffness is generally under predicted by all models.

In the bending-torsion load case (Fig. 7) again all the models predict bending displacement well, while the torsional stiffness is again under predicted. The shell/solid MSC PASTRAN model and the ANSYS model with pre-integrated matrix input are however the closest to the experimental results.

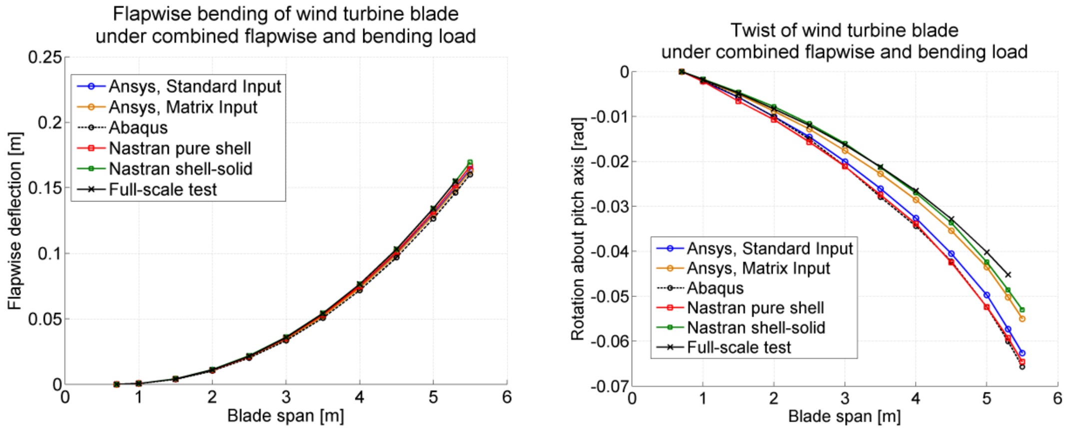


Figure 7. Results of the bending-torsion load case. 15 kN force + 10.43 kNm moment.

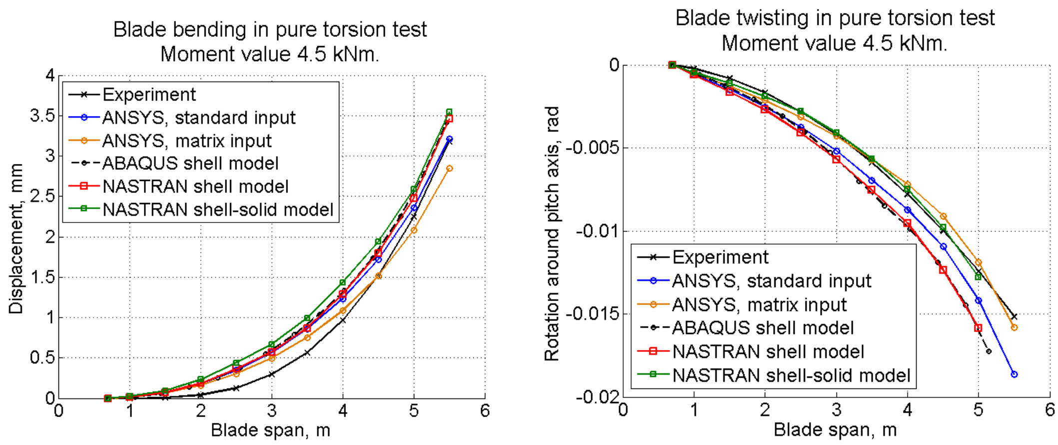


Figure 8. Results of the pure torsion load case. Torsional moment is 4.5 kNm.

In the pure torsion load case (Fig. 8) all the models over predict bending displacement in the root area, but satisfactory predict in the tip region, while the torsional stiffness is still under predicted, but again with the shell/solid MSC PASTRAN model and ANSYS model with pre-integrated matrix input closest to the experimental results.

5 Discussion

It is clearly seen from the figures 6-9 that the bending displacements are relatively well predicted in all load cases, while there is a large error in prediction of the twist angles. Over prediction of the twist angles range up to 31% for pure shell FE models.

One can also observe that the pure shell models share almost the same results in each load case and the same error in prediction of the torsional behaviour of the blade. Apparently the offsetting from the outer surface has a general problem with the prediction of torsion. This problem therefore does not depend on the solver, but is a general problem of the offset techniques. Probably it is because of wrong distribution of the shear flow around the cross-section when using offset, which leads to under estimation of the structural torsional stiffness and the twist angles accordingly.

The most successful model is still the MSC NASTRAN shell/solid model. The maximal error in prediction of the twist angle in all the load cases is 6.5%.

The pure shell ANSYS model with pre-integrated matrix input showed quite interesting results. The error in prediction of the twist angles did not exceed 9% in all the load cases. It is therefore evident that the use of pre-integrated matrix input does not have the same problem of under prediction of the torsional stiffness as in remaining pure shell models based on offsetting. At the same time the pre-integrated matrix input eliminates the possibility of accessing stress-strain distributions inside the shell elements. This is the main shortcoming of the approach, which highly reduces the attractiveness for using it in detailed FE analysis, where localised effects are wanted. However, for general structural behaviour purposes it seems to be highly attractive.

6 Conclusions

Erroneous prediction of the torsion behaviour of composite structures by pure shell FE models with offset to the outer surface exists and apparently it is a problem related to the offset techniques applied in most commercial FE codes. This fact makes these conventional pure shell models with offset to the outer surface undesirable for structural load cases where the composite structure undergo torsion deformations.

At the same time, the pure shell FE model with pre-integrated matrix input does not share the same problem. It was shown that satisfactory results in prediction of torsion behaviour with error not higher than 9% can be achieved. However, the fact that the stress-strain distribution in the shell elements is not available is a shortcoming of the pre-integrated input approach if such detailed results are desired, making a hybrid shell/solid FE model the best choice for modelling of composite structures with a high degree of complexity undergoing torsional deformations. However, for general structural behaviour purposes, where only the general structural deformations are needed, a pure shell FE model with pre-integrated matrix input seems to be highly attractive, due to the limited computational effort.

7 Acknowledgements

Vestas Wind Systems A/S provided the blade sections presented in this study. The support is gratefully acknowledged. The work is partly supported by the Danish Energy Authority through the 2007 Energy Research Programme (EFP 2007). The supported EFP-project is titled “Anisotropic beam model for analysis and design of passive controlled wind turbine blades” and has journal no. 33033-0075.

8 Literature

Last paragraph of your manuscript. Template Standard. Please use following scheme:

[1] Laird, D.L., Montoya, F.C., Malcolm, D.J., “Finite Element Modeling of Wind Turbine Blades” *Proceedings of AIAA/ASME Wind Energy Symposium*, Reno, Nevada, USA, AIAA-2005-0195, pp 9-17, 2005.

[2] Branner, K., Berring, P., Berggreen, C., Knudsen, H., “Torsional Performance of Wind Turbine Blades – Part II: Numerical Validation”, *16th International Conference on Composite Materials*, 8-13 July, Tokyo, Japan, 2007

[3] Berring, P., Branner, K., Berggreen, C., Knudsen, H., “Torsional Performance of Wind Turbine Blades – Part I: Experimental Investigation”, *16th International Conference on Composite Materials*, 8-13 July, Tokyo, Japan, 2007

[4] Dimitrov N., “Methods for Analysing Torsion Stiffness of Wind Turbine Blades”. *Master’s thesis*, Department of Mechanical Engineering, Technical University of Denmark, June, 2008.

[5] Berring P., Knudsen, H., “Torsional Behaviour of Large Wind Turbine Blades – Numerical and Experimental analysis”. *Master’s thesis*, Department of Mechanical Engineering, Technical University of Denmark, September, 2006.

[6] Sutton, M. A., Orteu, J-J., Schreier, H., “Image Correlation for Shape, Motion and Deformation Measurements”. Springer, 2009.

[7] Zenkert D., Battley M., “Foundations of Fiber Composites”. KHT, Stockholm, 2003.